ADRE® for Windows - instrumental in solving a complex vibration problem on a boiler feedwater pump

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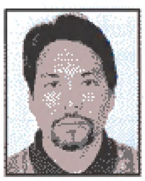

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For almost twelve years, the Vibration Group of the Operations Engineering Department (OED) has used the Bently Nevada ADRE 2 (Automated Diagnostics for Rotating Equipment) as a vibration data management and diagnostic tool. OED is part of the Operations Organization of Saudi Consolidated Electric Company in the Eastern Region of the Saudi Arabian Kingdom. The Department provides technical support services in all power transmission and generation stations, including machinery condition monitoring and diagnostics. ADRE was valuable in vibration data management and analysis and instrumental in the resolution of complex, as well as simple, vibration problems.

The ADRE 2 system was replaced by the updated ADRE for Windows and the DAIU 208 instruments. The new system provides more functions and flexibility that are essential in conducting routine and comprehensive analysis of various machinery vibration problems. ADRE for Windows was used to solve a complex case on a major auxiliary machine at one of the generating plants.

The case

The plant operates four steam turbine generators that supply 1600 MW to the power grid in Saudi Arabia. The vibration monitoring and analysis on the auxiliary machines are primarily done by the in-house vibration group with a portable vibration analyzer. Data analysis and archiving are done with the diagnostic software program provided with the analyzer.

During a routine survey on the auxiliary machines in Unit 4 on September 5, 1996, high vibration amplitudes were detected on the pump outboard (POR) and inboard (PIR) bearings on the Main Boiler Feedwater Pump. The vibration problem apparently developed after the pump's seal water sleeve was replaced because of excessive leakage. According to the plant's vibration group, the vibration amplitude on the pump outboard bearing vibration increased to 254 μm (10 mil) pp when the machine attained full speed.

In an attempt to correct the problem, Plant Maintenance replaced the pump bearings and realigned the machine train. However, the vibration levels did not decrease.

The vibration problem remained unresolved due to the limited capability of the in-house vibration group to acquire additional data for analysis. The OED Vibration Group conducted a survey and performed a comprehensive analysis on the vibration problem.

The 4-stage Delaval pump is driven by a steam turbine operating at 5500 rpm through a retractable

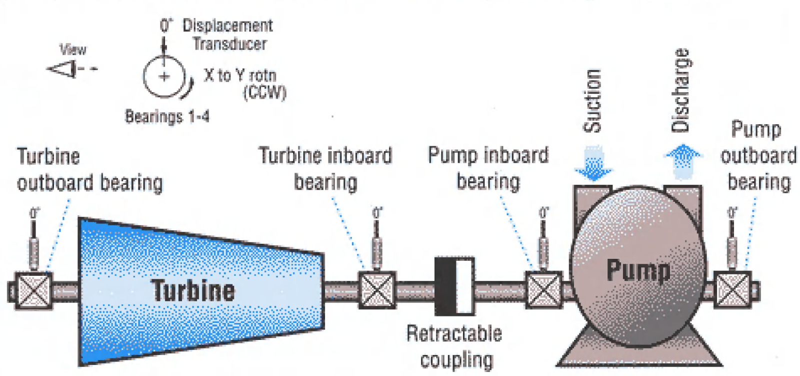

Figure 1. Unit 4 main boiler feedwater pump machine train configuration.

> *"Other information from shaft average centerline, orbit and full spectrum plots, that could have explained the pump rotor behavior during loading operation, was not available from the single probe installed on each bearing."*

coupling (Figure 1). The pump suction flow (machine load) is controlled by adjusting the turbine speed. Proximity probes are mounted on the bearings in the vertical direction and connected to the Bently Nevada vibration monitors. Figure 1 shows the basic configuration of the Main Boiler Feedwater Pump.

The Bently Nevada ADRE for Windows and the DAIU 208 (Data Acquisition Interface Unit) were used to gather vibration data from the proximity probes during the startup and shutdown. A portable vibration analyzer was used to acquire seismic vibration data on the bearings at full speed for additional information.

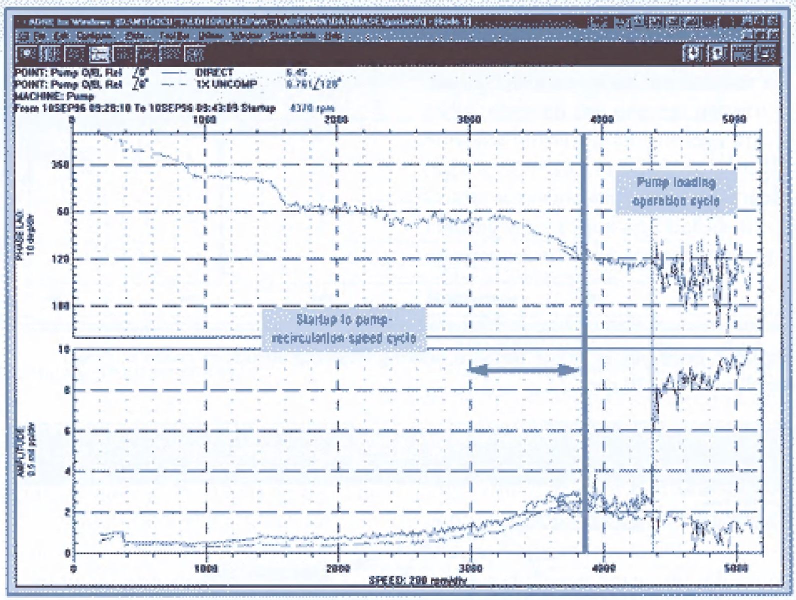

Figure 2. Bode plot of pump outboard bearing showing abrupt increase in direct amplitude at 4370 rpm.

Machine behavior

The initial survey on September 7, 1996, showed no adverse vibration behavior during the startup and loading cycle. The highest amplitude noted was 50 µm (1.96 mil) pp on the pump outboard bearing. When the OED Vibration Group left the plant, the machine was operating normally. However, during an ensuing startup, after the machine was shut down for a pending maintenance job, the vibration problem reappeared with an extremely high amplitude level (greater than 250 µm [10 mil] pp) on the pump outboard bearing. Another survey was made on September 10, 1996, to determine what happened.

The survey still showed normal vibration behavior and acceptable amplitude levels during the startup to the pump recirculation speed at 3800 rpm. This time, at approximately 4400 rpm, an abrupt

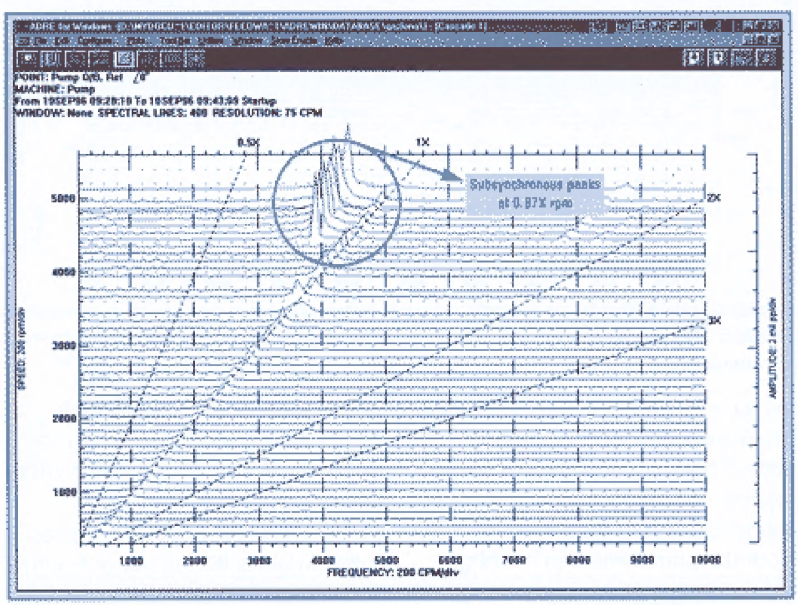

Figure 3. Half spectrum cascade plot of pump outboard bearing showing the subsynchronous peaks at 0.87X.

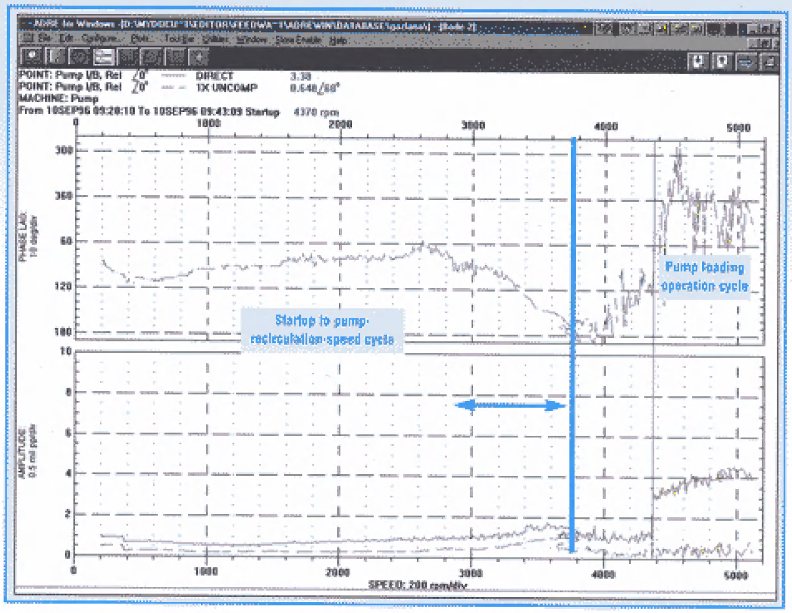

Figure 4. Bode plot of pump inboard bearing showing abrupt increase in direct amplitude at 4370 rpm.

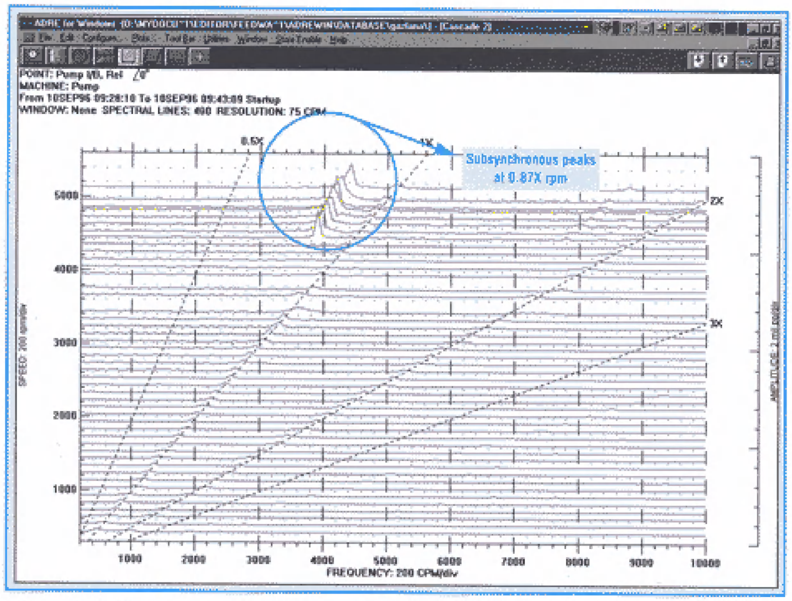

Figure 5. Half spectrum cascade plot of pump inboard bearing showing the subsynchronous peaks at 0.87X.

increase in vibration amplitude was observed on the pump outboard and inboard bearings during pump loading operation. An abnormal sound that seemed to originate from the pump was also heard during the same period. The pump's seal water temperature increased from a normal operating temperature of 38 °C (100 °F) to a high 91°C (195 °F). The seal water control became very erratic and eventually uncontrollable.

The behavior of the pump operating variables during pump loading operation is abnormal and clearly influenced the pump's vibration behavior and amplitude level.

The reason for the abnormalities could not be accurately determined during the time of the survey; therefore, additional information regarding the pump operation and internal components was gathered from the plant.

Analysis

The Bode plots (Figures 2 and 4) illustrate the abrupt increase in direct vibration amplitudes on the pump outboard and inboard bearings at 4370 rpm during pump loading. The half spectrum cascade plots (Figures 3 and 5), on the other hand, show subsynchronous peaks at 0.87X rpm, which were mainly responsible for the remarkable increase in vibration amplitudes at the pump bearings. Similar vibration signatures and patterns were also observed on the turbine bearings, but with lower amplitude levels. This vibration behavior was considered the result of the extremely high vibrations on the pump and was, therefore, not included in the analysis.

The vibration problem appeared to be complex and heavily influenced by underlying sources that were closely associated with the pump loading operation. Moreover, the exact nature of the problem could not be accurately determined from the available information. Other information from shaft average centerline, orbit and full spectrum plots, that could have explained the pump rotor behavior during loading operation, was not available from the single probe installed on each bearing.

Nevertheless, the information deduced from the Bode and half spectrum cascade plots provided valuable leads to the source of the problem.

Conclusions

The pump vibrations and behavior phenomenon that occurred during the loading operation were clearly incited by abnormalities in the pump operating variables. The

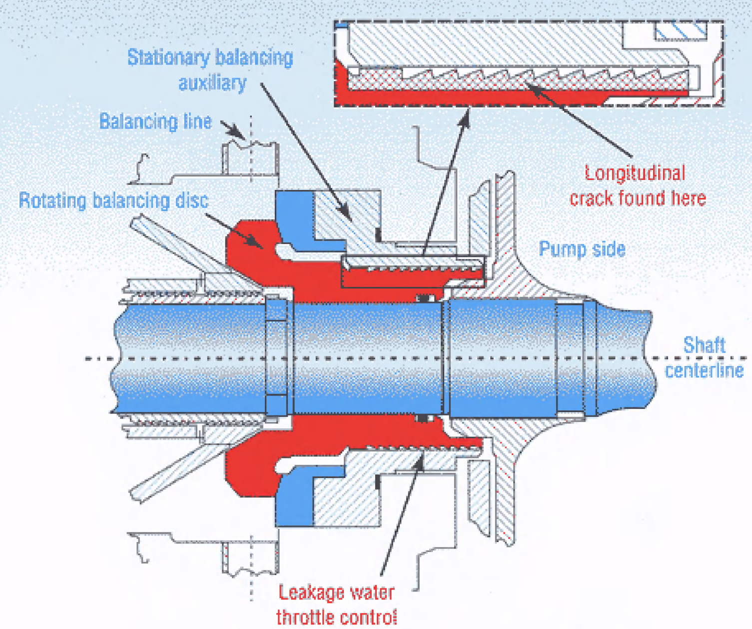

Figure 6. Cutaway view of balancing drum assembly.

source of the abnormalities could have been defective pump components responsible for maintaining rotor stability and fluid flow balance during operation.

The information gathered from the plant provided a clear understanding on the pump operation and internal components and gave an insight into the source of the problem. It was understood that the pump rotor stability and fluid flow balance during operation were primarily maintained by the balancing drum assembly. A cutaway view of a balancing drum assembly (Figure 6) shows the different components responsible for the control process.

Corrective measures

An inspection of the pump internals revealed a longitudinal crack in the rotating balancing disc in the balancing drum assembly. The entire pump rotor assembly was replaced instead of just the balancing drum because of concern about the integrity of the other internal parts on the existing rotor.

The crack in the balancing disc resulted in the loss of throttle control of pump water leakage to the balancing chamber. The excessive pump water in the balancing chamber contaminated the seal water. The contamination rendered the seal water control inoperable because the seal water temperature reached the saturation point of the controller, 93 °C (200 °F). The seal water control was essential in maintaining the pump rotor stability and fluid flow balance during operation.

Observations after correction

The machine behavior and vibration signatures observed in the survey conducted after replacement of the pump rotor assembly were normal. The Bode plots on the pump bearings (Figures 7 and 8) obtained during the survey on September 16, 1996, showed the normal pattern and low vibration amplitudes during the startup to load condition. Pump abnormalities seen during loading operations and before the correction were no longer present.

Summary

The subsynchronous peaks in Figures 3 and 5 were first thought to be an oil whip phenomenon on the pump bearings. However, the machine uses tilting pad type bearings, which are normally not susceptible to oil whirl and oil whip induced vibrations. Besides, oil whip signatures normally manifest at less than 0.5X (2600 cpm in this case) and not at 0.87X (4370 cpm),

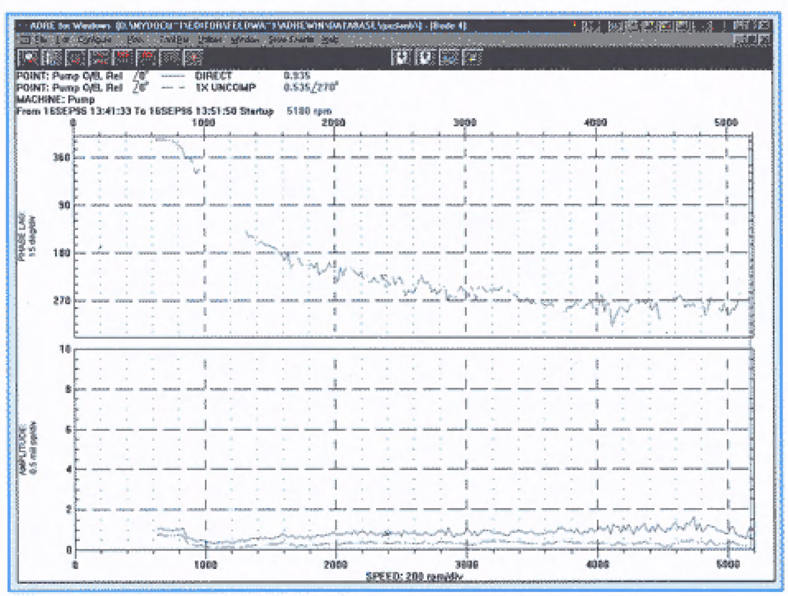

Figure 7. Bode plot of pump outboard bearing after replacement of pump rotor assembly.

as is seen on the vibration signatures. Also, while the whip frequency locks into the high eccentricity balance resonance frequency, this frequency tracks (stays proportional to) the increasing speed, a characteristic of whirl. While oil whirl typically occurs at less than 0.5X, pumping whirl can occur in the range of 0.8X to 0.9X, due to the increased fluid circumferential average velocity generated by the impeller. If the pump water leakage had somehow increased the fluid circumferential velocity in the seal, then the seal might have been the source of the instability.

With only the limited information provided by the single transducer at each bearing, the identity and source of the malfunction is extremely difficult to confirm, but the unstable flow in the seal water is a likely source of the instability.

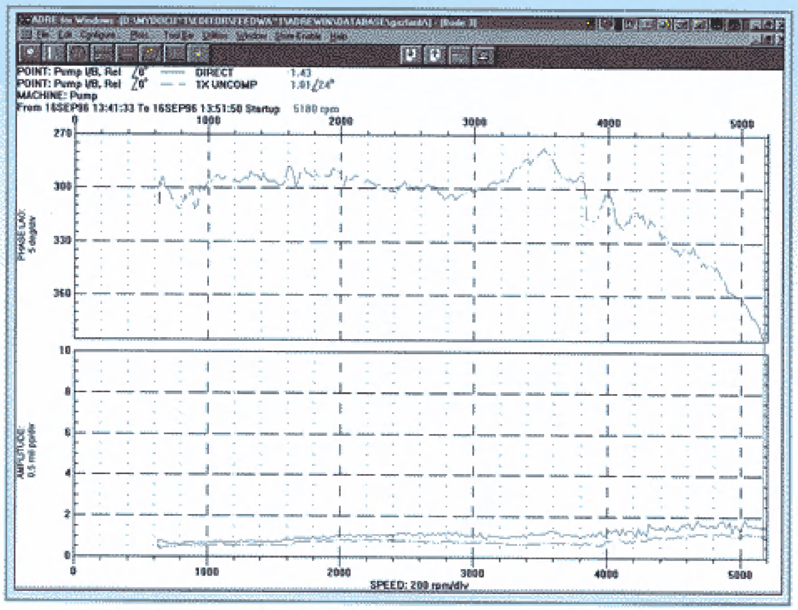

Figure 8. Bode plot of pump inboard bearing after replacement of pump rotor assembly.

Remarks

The problem on the Main Boiler Feedwater Pump was an extraordinary case and difficult to diagnose because of the underlying sources that heavily influenced the machine vibration behavior. The portable analyzer used by the in-house vibration group provided insufficient information for a conclusive analysis. Vibration information obtained from the single probe on each bearing also offered a limited view of the machine vibration behavior, particularly during loading. Nevertheless, the information deduced from the available plots produced by ADRE for Windows provided valuable leads to the source of the problem.

0.87X vibration revisited

As stated previously, the use of only one proximity probe at each measurement plane precludes the use of full spectrum, shaft orbits, and shaft centerline position information. The precession of the shaft orbit cannot be determined. This severely limits the diagnosis of the root cause of the problem. If the orbit precession and shape is *assumed* to be *forward* and *circular*, the following explanation is offered for the observed behavior of the rotor.

The Bode plot shown in Figure 2 indicates that the pump has a balance resonance frequency occurring at 3790 rpm. This is unusual in that most pumps, due to the hydraulic stiffening influence of the seals, typically have their balance resonance frequency above operating speed. The system stiffness of a pump is determined by the rotor geometry, seals, and bearings. If the stiffness properties of any of the elements is degraded or weakened, the resonance frequency will be lowered. It was noted by inspection that the rotating balance disk had a longitudinal crack which caused excessive leakage of the pump water to the balancing chamber. This also has the effect of lowering the stiffness of the system.

During the pump loading cycle, a subsynchronous vibration occurred at 0.87X. The pump speed at which this occurred was 4370 rpm. The source of the subsynchronous vibration is the excitation of the resonance frequency ($4370 \times 0.87 = 3801$ cpm). As the pump speed was increased from 4370 to above 5000 rpm, the subsynchronous vibration remained at 0.87X. As the pump is loaded, additional stiffening occurs from internal seals and wear rings. This increases the resonance frequency and maintains the excitation at 0.87X.

Upon completing repairs to the pump, the pump was restarted on September 16, 1996. The Bode plot shown in Figure 7 indicates satisfactory operational vibration levels. It further indicates the absence of a balance resonance at 3790 rpm.